Design of a Mixed-Mode MR Damper with Reduced Off-State Damping

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I would like to thank my family, who is everything to me.

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Abstract

Magnetorheological (MR) fluid, a suspension of iron particles in oil, is a 'smart fluid' whose yield stress changes with the application of a magnetic field. MR fluid is used in dampers to provide variable damping to systems. A damper in mixed mode, a common mode of damper design, consists of a piston in a cylinder with the surrounding volume filled with MR fluid. The critical parameter in this design is the gap distance between the piston and cylinder. An advantage of mixed mode dampers is that they are capable of producing very high damping forces (when a strong magnetic field is applied). However, they also have very high off-state force (when no magnetic field is applied).

The goal of this research is to lower the off-state damping forces. This is achieved by placing semi-circular cutouts around the piston circumference. Cutouts lower the off-state damping force and still maintain a high on-state holding force by varying the gap distance between the piston and the cylinder, therefore allowing more fluid to flow through in the off state. The damping forces were modeled using the Bingham plastic flow model. The design of an MR damper with cutouts and the advantages that cutouts provide were then analyzed. The size and numbers of cutouts were varied. Trends of dampers with cutouts were studied. Based on the analysis, a prototype damper was designed, built, and tested.

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1. Introduction/Motivation

1.1 MR Fluid

Magnetorheological (MR) fluid is a type of smart fluid, meaning that its rheological properties are capable of changing. Rheological properties are properties having to do with the deformation and flow of matter, including elasticity, plasticity, and viscosity. In the case of MR fluid, the yield stress of the fluid (the point after which the fluid flows freely) can go from zero (which indicates a Newtonian fluid) to some finite value (which indicates a non-Newtonian fluid). This change in rheological properties is induced by application of a magnetic field to the fluid. The change is completely reversible and occurs within milliseconds. Another type of smart fluid is electrorheological (ER) fluid. ER fluid behaves similarly to MR fluid, except that its property change is induced by an electric field rather than a magnetic field. Though the two fluids are comparable in the trends of their behavior, MR fluid surpasses ER fluid in many regards. For typical values of yield stress, the volume of ER fluid required is 100-1000 times that of MR fluid for similar performance. This means that MR-fluid devices can be designed to be much smaller than ER-fluid devices. Also, ER fluid is much more sensitive to temperature and contaminants than MR fluid is [1].

MR fluid consists of a carrier fluid, which is usually water or some type of oil, with micron-sized ferrous particles suspended in it. The ferrous particles usually compose 20-40% of the volume of the fluid [2]. Commercially available MR fluid also contains some type of additives to keep the particles suspended in the fluid. When there is no applied magnetic field, the fluid flows freely just as the carrier fluid normally would. The fluid is a Newtonian fluid, and flows when a shear stress is applied. The flowing fluid follows the equation $t = \mu ?$, where t is the shear stress, μ is the viscosity, and $?$ is the shear strain rate. This is referred to as the off state of the fluid. When a strong magnetic field is applied, the ferrous particles in the fluid align themselves with the magnetic flux lines, significantly increasing the fluid's resistance to flow. The fluid does not flow until the shear stress applied reaches a threshold value, which is the yield stress of the fluid, after which point the fluid flows freely. The nonzero yield stress is what makes the fluid in this state one class of a non-Newtonian fluid. This fluid can be described by the equation $\tau = \mu\dot{\gamma} + \tau_y$, where τ_y is the yield stress of the fluid. As can be seen from the equation, once the shear stress surpasses the yield stress of the fluid, the fluid flows freely. This is referred to as the on state of the fluid. Aside from simply being off or on, the MR fluid can obtain a yield stress anywhere in that range by varying the strength of the applied magnetic field. The response time for the fluid to change its properties once a field is induced is on the order of milliseconds.

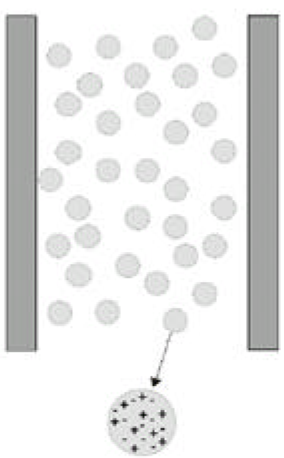

Figure 1. Off state of fluid, from [3]

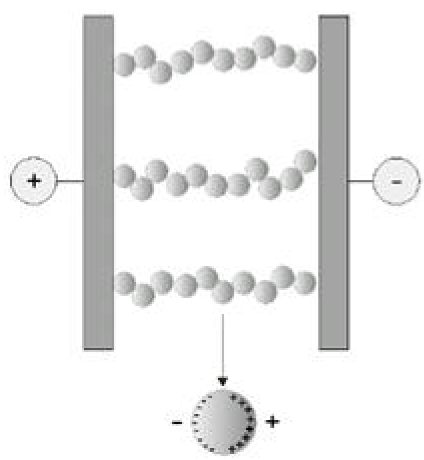

Figure 2. On state of fluid, from [3]

1.2 MR Fluid Dampers

MR fluid is currently used in commercially produced dampers for many applications, a few of which are described in the next section. MR fluid is useful in dampers because it is quiet with a quick response time, and it has the ability to facilitate variable damping. Variable damping can be desirable because the damping can adjust in real time to different loading conditions and vibrations. Variable damping has previously been achieved in mechanical systems by varying the orifice size. However, the response time of the MR dampers is much quicker than that of a damper whose orifice size changes [4].

There are two main modes by which MR fluid can operate in MR-fluid devices. In both modes, the fluid is between two plates that serve as the poles for the magnetic field. The first mode is known as direct shear mode (Figure 3). In this mode, the plates are moving relative to each other. Shear mode dampers are typically used for low-force applications and are generally made with a sponge that is saturated in MR fluid rather than free-flowing fluid. The second mode is flow mode, also known as valve mode (Figure 4). In this mode, the plates are stationary, and pressure drives the fluid flow. Flow mode dampers (Figure 5) are capable of producing very high forces, but are generally difficult to design. A third mode, mixed mode, is a combination of the shear and flow modes. Mixed mode dampers, like flow mode dampers, are capable of producing large forces, only the design is much simpler. A fourth mode which is not used very often is known as squeeze-film mode.

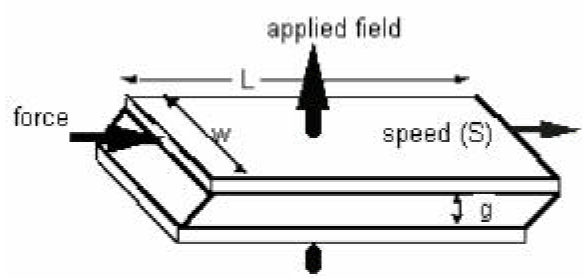

Figure 3. Direct shear mode, from [5]

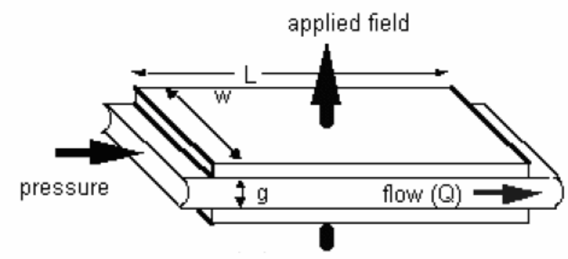

Figure 4. Pressure driven flow mode, from [5]

A mixed mode damper (Figure 6) consists of a piston encased in a cylinder. The volume between the two components is filled with MR fluid. As the piston moves relative to the cylinder, the fluid moves in the opposite direction through the annulus formed by the piston and the cylinder. This makes the area of the thickness of the annulus, or the gap between the piston and the cylinder, the critical parameter in this design. Coils are wrapped around the piston to produce the magnetic field. When the magnetic field is induced, the iron filings in the activated fluid between the piston and the cylinder align themselves with the magnetic field, significantly increasing the apparent viscosity of the fluid in the annulus.

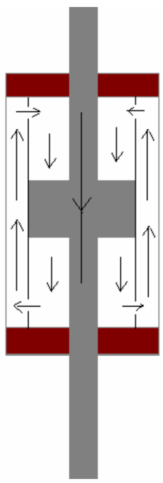

Figure 5. Flow mode device, adapted from [6]

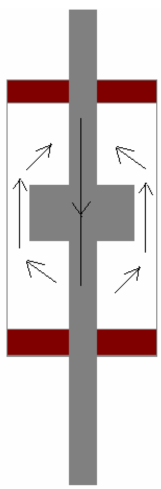

Figure 6. Mixed mode device, adapted from [6]

1.3 Existing applications of MR dampers

One of the most commercially successful MR damper applications is in heavy vehicle seating. Incorporating an MR damper into the design of the seat (Figure 7), makes the ride much more comfortable and safe for the driver than traditional suspended seats. The seat automatically adjusts to the driver's weight and continually changing road conditions.

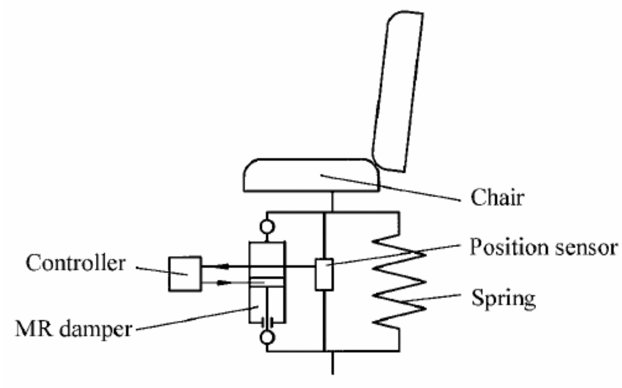

Figure 7. Structure for an MR semi-active controlled damper suspended seat, from [7]

Another application of MR dampers is in larger scale damping such as in bridges and buildings. The Dongting Lake Bridge in China is the world's first cable-stay bridge to incorporate MR dampers [8]. For several months each year in China, severe wind and rain were often causing all 156 cables on the bridge to vibrate at the same time. Realizing the potential hazard, new cables were designed to reduce the vibrations. The new cables each have two MR dampers supporting them (Figure 8). The design of the dampers had to be modified slightly in order to endure the weather, but the basic principles of the damper remained the same. The MR dampers work well because they can provide large damping forces even with just slight displacement of the piston, and because the different cable lengths are not a problem for the adjustable damping of an MR damper.

earthquakes [11].

Figure 8. Bridge in China with MR dampers, from [8]

Another structural application of MR dampers is in buildings for damage control against earthquakes (Figure 9). MR dampers surpass other damping systems because of many reasons, including the large dynamic range, quick response time, small size, large temperature range, and mechanical simplicity [2]. Also, since an MR damper is still a passive damper even without a power source, it is fail-proof, which is essential considering aftershocks that occur with an earthquake.

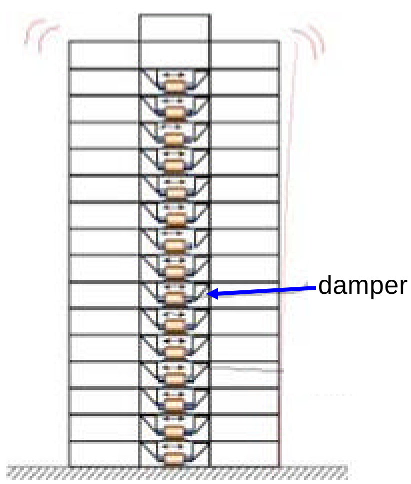

Figure 9. A building protected against strong vibrations with MR dampers, from [2]

Other applications for which MR dampers are currently used include gun recoil control systems [9], car seat safety belts [2], prosthetic limbs [10], and vibration control in helicopters [11].

1.4 Motivation for this project

The objective of this research is to design an MR fluid damper with a larger dynamic range that has previously been achieved. Specifically, the goal is to lower the off-state force. The damper will be mixed mode because this mode is commonly used for its capability of large damping forces in the on state, but can be somewhat limiting by its high off-state damping forces. As stated previously, the most critical parameter in the design of a mixed mode damper is the gap distance between the piston and the cylinder. The smaller the gap, the less fluid is allowed to flow through, therefore increasing the force of the on state, but also increasing the force in the off state. Increasing the gap width allows more fluid to flow through, decreasing the force in the off state, but also decreasing the force of the on state. The challenge is to optimize the geometry of the damper, mainly the distance between the piston and the cylinder, such that the achievable force range fits the application. Unfortunately, the achievable force range of a typical mixed mode damper is not large enough for many applications in which small off-state damping is desirable. To address this challenge, a new design of a damper was proposed in order to reduce the force in the off state while maintaining the high force in the on state. The idea is to contour a piston such that it is no longer a simple cylinder, but rather a cylinder with cutouts along the circumference (Figure 10). The cutouts provide extra room for fluid to flow through, decreasing the force in the off state without much reduction of force in the on state.

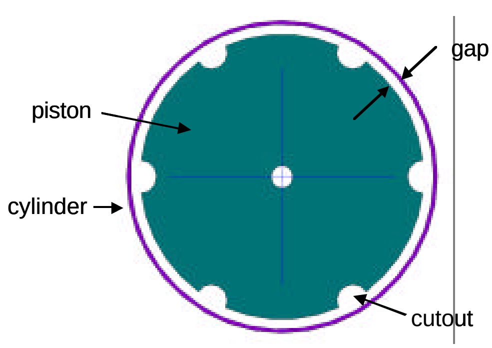

Figure 10. Cross-sectional view of a damper with a contoured piston

1.5 Overview

The remainder of this thesis will detail the work on designing an MR damper with a low off-state damping force. Chapter 2 addresses the theory behind the design. In Chapter 3, the design of the damper is analyzed based on the model. Design parameters, constraints, and trends are examined in detail. Chapter 4 presents the details behind the design of a prototype damper. Chapter 5 concludes this work and contains recommendations for future work.

2. Theory

Modeling of the fluid flow in a regular mixed mode damper has been done and validated by experiment [12]. By examining the flow of an MR fluid through a certain damper geometry, damping forces can be predicted for various conditions. By just focusing on the force produced by the flow of the fluid, it is assumed that the damper components are massless and that the force of friction between non-fluid components is negligible compared to the force of the flow. The derivation of the force for a contoured-piston damper is similar to that of a regular-piston damper. The difference is that with a regular piston the gap distance is constant, whereas with a contoured piston the gap distance varies. For this reason, additional assumptions need to be made when modeling the flow. The contoured piston in this project is a cylindrical piston with semi-circular cutouts around the piston circumference.

To model the flow of the MR fluid inside the damper, the Bingham plastic flow model was used. This model represents fluids such as MR fluid that act as semi-solids at low stresses, but after the threshold of stress (the yield stress) is reached, the fluid flows freely as a normal viscous Newtonian fluid. Other than MR and ER fluids, mud flow is often modeled by Bingham plastic flow in offshore engineering [13].

MR fluid is classified as a non-Newtonian fluid. The basic principle of a Newtonian fluid is that the fluid flows when a shear stress is applied. The flowing fluid follows the equation $t = \mu ?$, where t is the shear stress, μ is the viscosity, and $?$ is the shear strain rate, which is equal to $\frac{du}{dy}$. Being a Bingham plastic fluid, which is one class of a non-Newtonian fluid, MR fluid does not flow until the shear stress applied reaches a threshold value, which is the yield stress of the fluid. The state of the fluid before it reaches the yield stress is known as the pre-yield phase. After this yield stress is reached, the state of the fluid is known as the post-yield phase, and the fluid flows as a normal viscous Newtonian fluid with the shear stress proportional to the shear rate. This post-yield phase is what the Bingham plastic model characterizes. This phenomenon can be described by the equation $t = \mu ? + t_y$, where $?$ is the shear rate, μ is the plastic viscosity, t is the shear stress, and t_y is the yield stress of the fluid. The yield stress of an MR fluid is highly dependent on the applied magnetic field, so varying the intensity of the field varies the yield stress of the fluid. MR fluids purchased through the Lord Corporation, the world's primary manufacturer of MR fluids, are accompanied by a fluid property chart that illustrates the relationship between the yield stress and the applied field. The general relationship is $t_y = a_2 H_2 + a_1 H + a_0$, where H is the magnetic field intensity, and the constants a_2, a_1, and a_0 differ for each fluid.

The Bingham plastic model is useful for modeling the flow of MR fluid, and has been proven to be accurate through flow visualization studies done by the Lord Corporation [12]. This model has certain assumptions and limitations that do not make it all encompassing of MR fluid flow in all situations and states. A very important assumption is that the fluid is in the post-yield phase (the yield stress has already been reached). In high force applications such as those for which the dampers examined in this project are designed, this assumption is valid. It is also assumed that fluid is flowing at a constant shear rate.

Analyzing the forces on the flow through the annulus of the damper, the governing equation comes from the Navier-Stokes equations. In force equilibrium, the simplified governing equation is

$$? \frac{\partial u}{\partial t} + \frac{\partial t}{\partial y} + \frac{t}{y} = \frac{\partial p}{\partial z} \quad (2.1),$$

where u is the velocity, y is the radial coordinate, z is the longitudinal coordinate, and p is the pressure developed due to the piston displacement. In this analysis, the equation is simplified since only quasi-steady flow is examined. Therefore, the fluid inertia term can be neglected. It is assumed that the pressure varies linearly across the gap, so the equation can be further simplified to

$$\frac{\partial t}{\partial y} = \frac{\Delta P}{L} \quad (2.2),$$

where L is the length of the piston.

Since the piston has cutouts in it, the distance between the piston and the cylinder is not constant along the entire piston. Areas along the piston circumference where there is not a cutout have a small gap that is a constant d_S. Areas along the piston circumference where there is a cutout have a larger gap, d_L, that ranges from d_S to $d_S + d_{\text{cutout}}$. This gap distance (Figure 11) can be described by the equation

$$d_L = \sqrt{xb_L - x^2} + d_S \quad (2.3)$$

where b_L is the arc length of the piston circumference that the cutout takes up and x is the distance along the circumference of the piston where $0 < x < b_L$ (Figure 11).

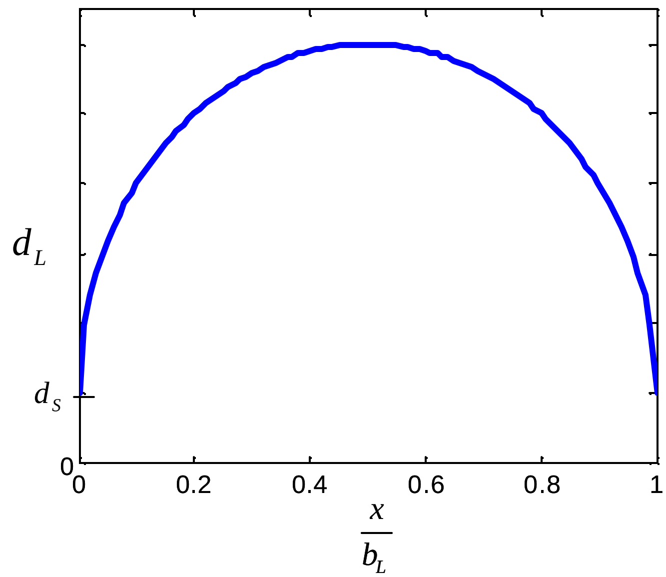

Figure 11. Gap width at a cutout

The yield stress of the fluid is dependent on the magnetic field strength on the fluid, which is a function of the geometry of the damper (Figure 12) and the current in the coils. A greater number of coils increases the magnetic flux density, but the number of coils, N, that is wrapped around the piston core is constrained by the amount of space there is. Coils are wrapped around the piston core in layers that extend to the piston end discs. With cutouts in the piston, the coils can only be wrapped up to the deepest point of the cutouts, so the allowed space for coils is reduced.

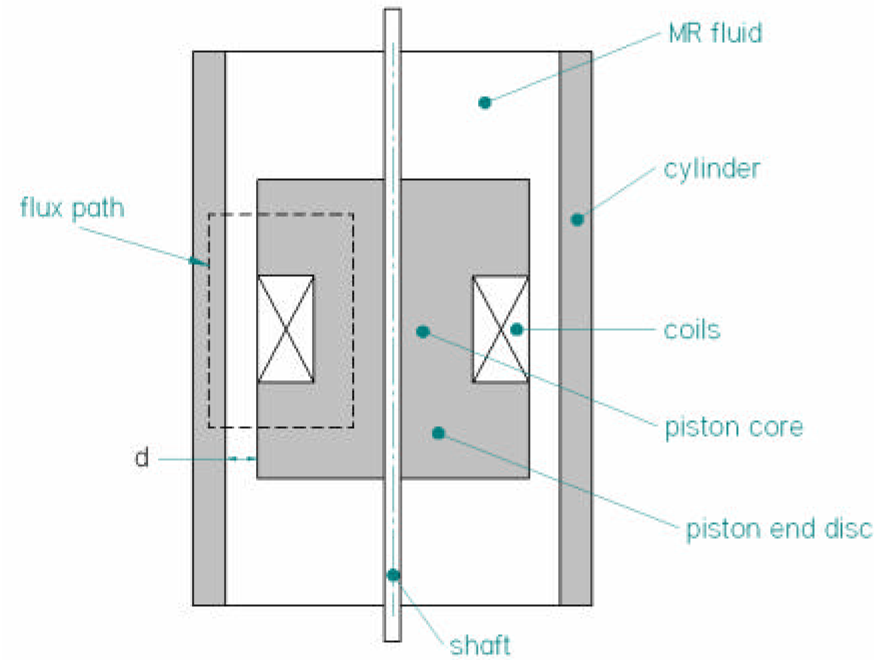

Figure 12. Flux path through the damper, adapted from [5]

The magnetic flux path, as seen in Figure 12, goes from the piston core, through the piston end disc, through the fluid, through the cylinder, through the other piston end disc, and back through the piston core. Where there is a cutout, the path through the piston decreases, and the path through the fluid increases.

A significant assumption made in this analysis is that the magnetic flux lines are equally distributed and traverse the gap along the radial line (Figure 13).

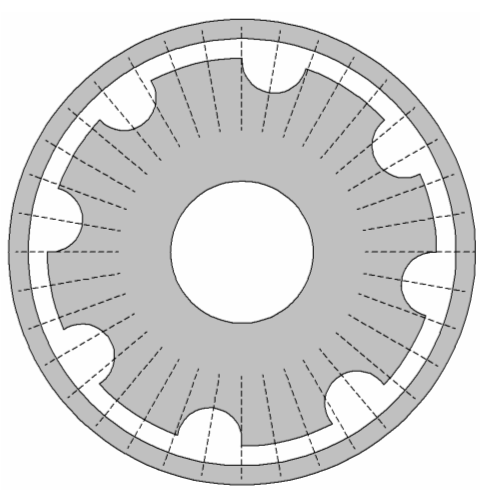

Figure 13. Assumed distribution of magnetic flux lines

This assumption is valid in a regular damper because the gap distance is constant, and the piston is the same in the radial direction all the way around. However, when there are cutouts, the magnetic flux tends to concentrate around sharp edges, making the magnetic flux unequally distributed, and not necessarily all in the radial direction. Here it is assumed that the flux lines all go in the radial direction and that the concentration of flux lines depends upon the gap at a particular point, but not upon the geometry of the entire gap.

The piston, fluid, and cylinder, are components in series that each exhibit a magnetic resistance known as reluctance. The reluctance of a component is given by the equation $R = \frac{l}{\mu A}$, where l is the mean path of magnetic flux lines in the component, μ is the permeability of the component, and A is the cross-sectional area of the component. The piston and the cylinder should be made of a material, such as low carbon steel, that has a fairly high permeability in order to produce a strong magnetic field. The permeability of the MR fluid is significantly lower than that of the material that should be used for the piston and cylinder, so increasing the fluid field path and decreasing the piston field path of the magnetic field will increase the reluctance. Since the components are in series, their reluctances (which are analogous to resistances in an electrical circuit) can be added together to find the total reluctance of the system.

The magnetic flux can then be calculated using the relationship $\phi = \frac{NI}{R_T}$, where I is the current in the coils, ϕ is the magnetic flux, and R_T is the total reluctance of the system. Using the magnetic flux and the area, the magnetic flux density can be calculated as $B = \frac{\phi}{A}$, where A is the cross-sectional area of the core of the piston around which the coils are wrapped. Ampere's Law relates the magnetic flux density, B, and the magnetic field intensity, H. The law states that the two values are proportional, or $B = \mu_r \mu_0 H$, where μ_0 is the permeability of free space and μ_r is the relative permeability of the medium through which the flux is traveling [14]. The yield stress, τ_y, is a fluid-specific property that is a function of the magnetic field intensity. The general relationship is $t_y = a_2 H_2 + a_1 H + a_0$, where the constants a_2, a_1, and a_0 differ for each fluid. When the gap is larger, as in the case where there is a cutout in the piston, the yield stress is lower because the magnetic field intensity is lower. The yield stress is constant in the areas with a small gap, but varies along the cutout in the areas with large gaps.

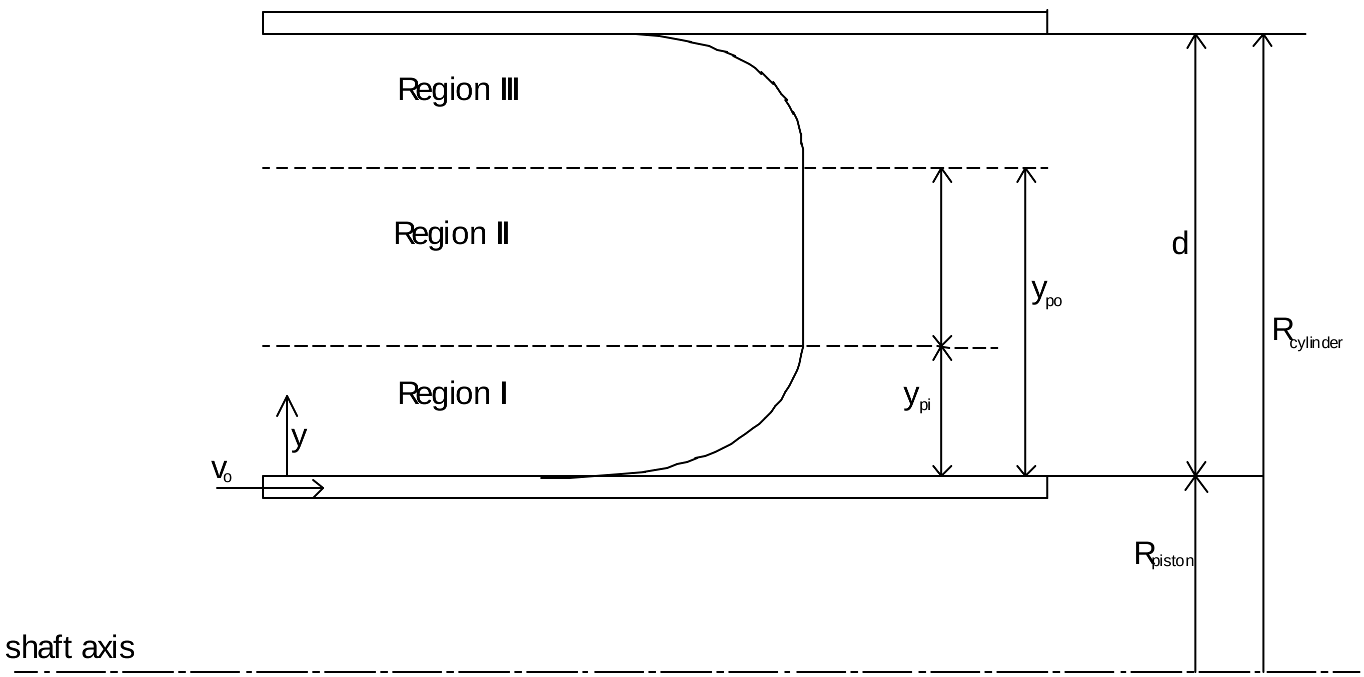

Figure 14. Velocity flow profile, adapted from [3]

When the fluid begins flowing through the annular region after the yield stress has been reached, three regions of flow develop (Figure 14), with d being the distance from the piston to the cylinder. In regions I and III, the fluid is in the post-yield phase, but in region II, the fluid is in the pre-yield phase and moves as a solid, which is known as plug flow. Regions I and III are similar, differing in that the plate in region III (the wall of the outer casing) is stationary, while the plate in region I (the piston) is moving at velocity v_o. As seen from Figure 14, regions I and II meet where $y = y_{pi}$, and regions II and III meet where $y = y_{po}$. The thickness of region II is the plug thickness, $d = y_{po} - y_{pi}$. It is assumed that regions I and III have the same thickness, or $y_{pi} = d - y_{po}$. This is a valid assumption in flow mode where the two plates do not move relative to each other, but even in a mixed mode with a contoured piston, the assumption should be reasonably valid because the velocity of the piston is negligible compared to the velocity of the fluid. The plug thickness varies with the flow conditions. When there is no magnetic field and the yield stress is 0, the plug thickness is also zero, and only regions I and III exist. However, when the strength of the magnetic field is fairly large, the plug dominates over the other regions of flow.

The equation for the shear stress in regions I and III is

$$\tau = \mu \frac{du}{dy} + \tau_y \quad (2.4),$$

where $\tau > \tau_y$. However, region II, where the fluid is not flowing, the equation for the shear stress is

$$\frac{du}{dy} = 0 \quad (2.5)$$

where $\tau < \tau_y$. Each region of flow is considered separately. As in the calculation of the yield stress, the flow in each region is also calculated separately for the small gaps and the large gaps. In the small gaps, the flow profile is constant across the gap, but for the large gaps, the flow profile continuously changes across the gap with the cutout because the width of the gap is changing.

Region I extends from the piston to region II. Therefore, the boundary conditions for this region are $u(0) = v_0$, where v_0 is the piston velocity, and $\frac{du}{dy}\left(y_{pi}\right) = 0$. For the small gaps, y_{pi} is constant, but for the large gap y_{pi} is a function of gap distance. Combining Equations 2.2 and 2.4 yields the equation

$$\mu \frac{d^2u}{dy^2} = \frac{\Delta P}{L} \quad (2.6).$$

Integrating this equation twice and using the given boundary conditions, the velocity of the flow through the gap can be determined as a function of y, $u_1 = \frac{\Delta P}{2\mu L}\left(y^2 - 2y_{pi}y\right) + v_o$.

It is known that at the boundaries of region II, the shear stress is equal to the yield stress, or $\tau\left(y_{pi}\right) = \tau_y$ and $\tau\left(y_{po}\right) = -\tau_y$. Integrating Equation 2.2 using these boundary conditions and solving for the shear stress gives the equations $\tau_y = \frac{\Delta P}{L} y_{po} + C_1$ and $-\tau_y = \frac{\Delta P}{L} y_{pi} + C_1$. Subtracting these two equations leads to the following expression

$$y_{po} - y_{pi} = \frac{2L\tau_y}{\Delta P} = d \quad (2.7).$$

Region III is very similar to region I, with the only difference being that the wall in region III (cylinder) is stationary, while the wall in region I (piston) is moving. Therefore, the boundary conditions for this region are: $u(d) = 0$ and $\frac{du}{dy}\left(y_{po}\right) = 0$. Now, integrating Equation 2.6 with these boundary conditions, the velocity of the flow in region III can be determined to be $u_3 = \frac{\Delta P}{2\mu L}\left(y^2 - d^2 + 2y_{po}(d - y)\right)$.

Next, using the fact that region II is the plug flow region where the velocity is constant, the flow velocity in this region can be determined from the fact that $u_2\left(y\right)=u_1\left(y_{pi}\right)=u_3\left(y_{po}\right)$. Using this known relationship, the velocity in region II can be determined to be

$$u_2=\frac{-\Delta P}{2\mu L}y_{pi}^2=\frac{-\Delta P}{2\mu L}\left(y_{po}+d\right)^2=\frac{\Delta P}{8\mu L}\left(d-\delta\right)^2.$$

For a damper in flow mode, the thicknesses of regions I and III are identical. Although this is not exactly true for a mixed mode damper, where the relative velocity of the piston and the cylinder would produce slightly different thicknesses for the regions, it can be assumed that the difference would be negligible. From this assumption, expressions for y_{pi} and y_{po} can be derived to be $y_{pi}=\frac{d}{2}\left(1-\frac{\mathrm{d}}{d}\right)$ and $y_{po}=\frac{d}{2}\left(1+\frac{\mathrm{d}}{d}\right)$.

Next, the volumetric flow rate in each region must be determined. The velocity in each region is expressed in terms of ΔP. Therefore the total flow rate can be expressed in terms of ΔP, because the flow rate of the fluid must be the amount of fluid displaced by the volume of the piston as it moves with a known velocity. That is

$$Q=A_p v_o \quad (2.8),$$

where Q is the total volumetric flow rate and A_p is the cross-sectional area of the piston. The volumetric flow rate in each region is determined by integrating the flow velocity over the region. This integration works out nicely in the areas of the small gap (denoted by a subscript S), where the gap is constant, and the integration of the flow is done with respect to y. However, in the areas with a large gap (denoted by a subscript L), where the gap is not constant, in addition to integrating with respect to y, it is necessary to integrate across the cutout, or with respect to x, because the gap changes with respect to x. Therefore,

$Q_{S1} = b_S \int_0^{y_{pi}} u_1(y)\,dy = nb_S \left(-\frac{d_S^3 \Delta P}{24\mu L}\left(1-\frac{\mathrm{d_S}}{d_s}\right)^3 + \frac{1}{2} v_o d_S \left(1-\frac{\mathrm{d_S}}{d_s}\right)\right)$, where b is the arc length of the segment and n is the number of segments there are around the piston circumference.

$$Q_{S2} = b_S \int_{y_{pi}}^{y_{po}} u_2(y)\,dy = -\frac{nb_S d_S^3 \Delta P}{8\mu L}\left(1-\frac{\mathrm{d_S}}{d_S}\right)^2, \quad Q_{S3} = b_S \int_{y_{po}}^{d} u_3(y)\,dy = -\frac{nb_S d_S^3 \Delta P}{24\mu L}\left(1-\frac{\mathrm{d_S}}{d_S}\right)^3$$

$$Q_{L1} = \int_0^{b_L}\int_0^{y_{pi}} u_1(y)\,dydx = \int_0^{b_L}\int_0^{y_{pi}} \left(\frac{\Delta P}{2\mu L}\left(y^2 - 2y_{pi}y\right) + v_o\right)dydx = \int_0^{b_L}\int_0^{y_{pi}} \left(\frac{\Delta P}{2\mu L} y^2 - d_L y\left(1-\frac{\mathrm{d_L}}{d_L}\right) + v_o\right)dydx$$

where $\mathrm{d}_L \neq \mathrm{d}_S$ and $d_L = \sqrt{xb_L - x^2} + d_S$.

Similarly, $Q_{L2} = \int_0^{b_L}\int_{y_{pi}}^{y_{po}} u_2(y)\,dydx = \int_0^{b_L}\int_{y_{pi}}^{y_{po}} \frac{\Delta P}{8\mu L}\left(d - \mathrm{d}_L\right)^2 dydx$, and

$$Q_{L3} = \int_0^{b_L}\int_{y_{po}}^{d} u_3(y)\,dydx = \int_0^{b_L}\int_{y_{po}}^{d} \frac{\Delta P}{2\mu L}\left(y^2 - d_L^2 + 2y_{po}\left(d_L^2 - y\right)\right)dydx$$

$$= \int_0^{b_L}\int_0^{y_{pi}} \left(\frac{\Delta P}{2\mu L} y^2 + d_L\left(d_L - y\right)\left(1+\frac{\mathrm{d_L}}{d_L}\right)\right)dydx .$$

The total volumetric flow rate is equal to the volume displaced by the piston as it moves with a known velocity, so $Q = Q_{S1} + Q_{S2} + Q_{S3} + Q_{L1} + Q_{L2} + Q_{L3}$, where Q is known from Equation 2.8. From Equation 2.7, d is known in terms of t_y and can be substituted accordingly. Also, it is known that the change in pressure is proportional to the force on the damper, that is

$$\Delta P = -\frac{F}{A_p} \quad (2.9).$$

Plugging this expression into Equation 2.8, a nonlinear expression is given in terms of force. Forces were computed using the Newton Raphson method in MATLAB.

In summary, the theory for modeling an MR damper consists of two divisions, the yield stress and the flow analysis. The yield stress is a function of the magnetic field strength, which depends upon the current in the coils and the geometry of the damper. The flow is analyzed using the Bingham Plastic flow model, which divides the flow into three regions. Regions I and III are in the post-yield phase, and the velocity profiles in these regions can be determined by combining and integrating Equations 2.2 and 2.4. Region II is in the pre-yield phase where the flow velocity is constant and independent of the y-direction. The velocity in this region can be determined from Equation 2.5 and known boundary conditions. The flow rate in each region can then be determined by integrating the velocity over the area of the gap. Summing the flow rates in each region gives the total flow rate in terms of ΔP, and is equal to the volume displaced by the piston as it moves with a known velocity. The known relation in Equation 2.9 is then used to replace ΔP for F in the flow rate equation, and a nonlinear equation for the force is the result. The force is solved for by the Newton Raphson method.

3. Design Analysis

3.1 Design parameters

In designing a mixed mode MR damper, there are many parameters that can be varied in order to obtain different results. Depending upon the desired application of the damper, some of these parameters may also be constrained to certain values. Even within those constraints though, there is an optimal design. For this project, the optimal design was considered that which obtained the lowest off-state damping force with a high on-state force.

The parameters in Table 1 are all of the geometric parameters that can be independently varied in the design of an MR damper. In addition to the geometric parameters, material parameters can also be varied. Depending upon the fluid that is chosen, the relationship between the magnetic field strength and the yield stress can vary significantly, as can the viscosity. The material of the other components can also be varied. The material of the piston and the cylinder must conduct the magnetic field, while the material of the shaft and insulator must be made of material that will not conduct the magnetic field in order to concentrate the field on the fluid. The permeabilities of the piston and the cylinder depend upon the materials chosen, and will affect the strength of the magnetic field.

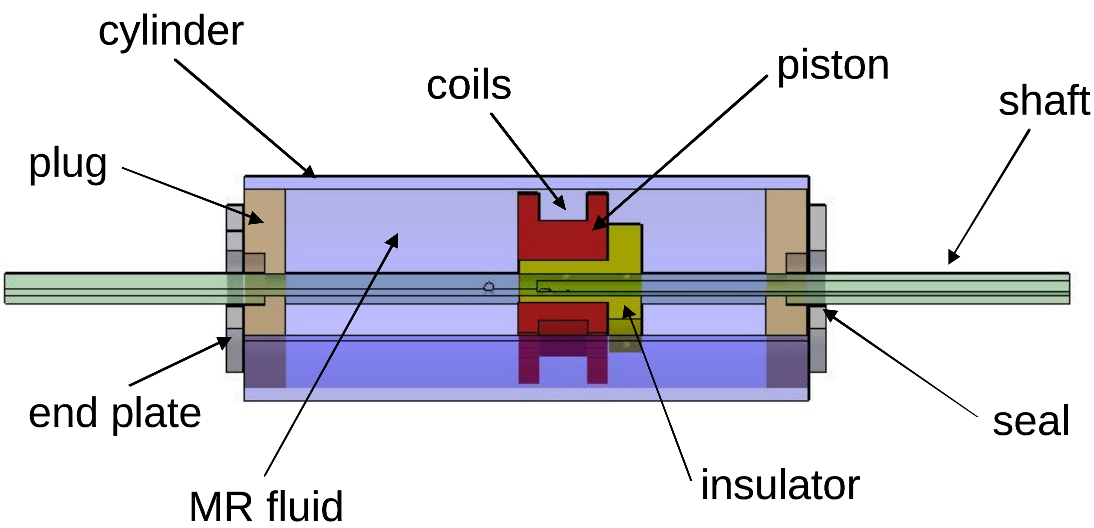

Figure 15. Mixed mode damper parts

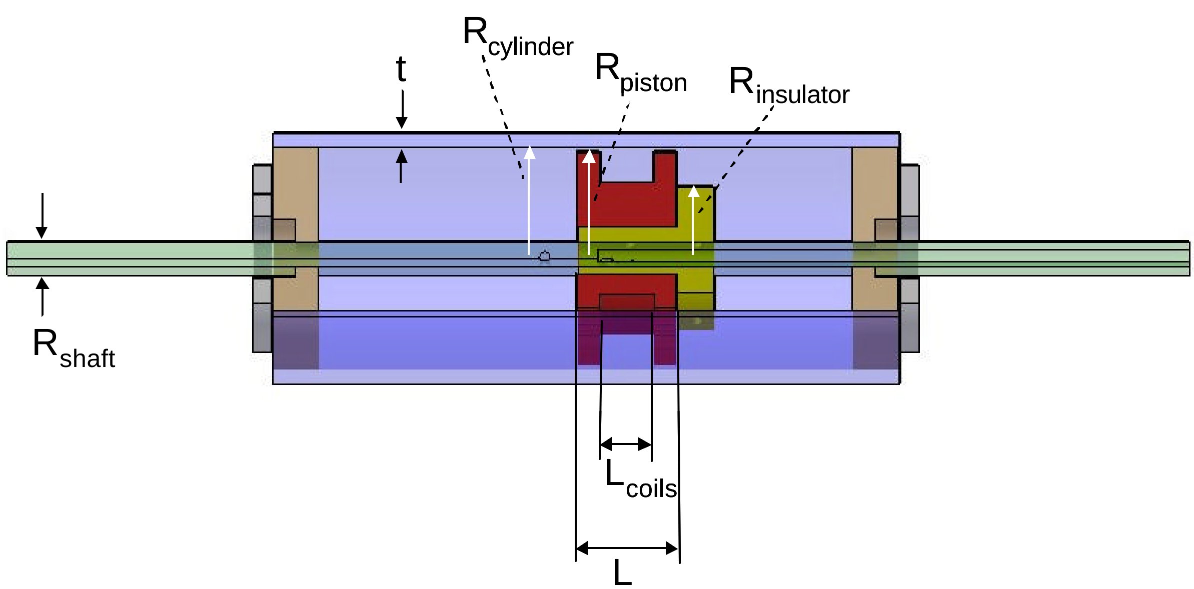

Figure 16. Mixed mode damper parameters

Table 1. Parameters of a mixed mode damper

Design Parameter	**Variable Name**	**Units**	**Varied/Constant**
Radius of piston	R_{piston}	Meters	Varied
Radius of cylinder	$R_{cylinder}$	Meters	Varied
Shape of cutouts		(Shape)	Constant
Number of cutouts	N	(Integer)	Varied
Size of cutouts	R_c	Meters	Varied
Radius of shaft	R_{shaft}	Meters	Constant
Diameter of wire	d_{wire}	Meters	Constant
Total length of piston	L	Meters	Constant
Length of piston without coils	$L_{noncoils}$	Meters	Constant
Length of piston with coils	L_{coils}	Meters	Constant
Radius of piston core	R_{pcore}	Meters	Constant
Thickness of cylinder	T	Meters	Constant
Number of sections of coils	N_S	(Integer)	Constant
Radius of insulator	$R_{insulator}$	Meters	Constant

For the analysis of the effect of the cutouts, only a few parameters in the table above were varied, but the effects of varying some of the other important parameters were also noted. This was done as a rough sensitivity analysis. The design parameters studied were the radius of cutouts, the total length of piston, the thickness of cylinder, the number of cutouts, the radius of piston, the thickness of piston core, and the number of turns. For each parameter, an operating point was chosen (Table 2) to be the median examined. The operating points for most of the parameters were the values in the prototype damper discussed in Chapter 4. However, for the number of cutouts and the number of turns, the value in the prototype damper was the maximum possible, so the value chosen for the sensitivity analysis was half of the value in the prototype damper. Each parameter was varied relative to the operating point, and its effect on the output force, both in the on and off states was determined. The percent changes in the forces were computed relative to the force provided when the parameter was equal to its operating point value. When each parameter was varied, all other parameters were kept constant.

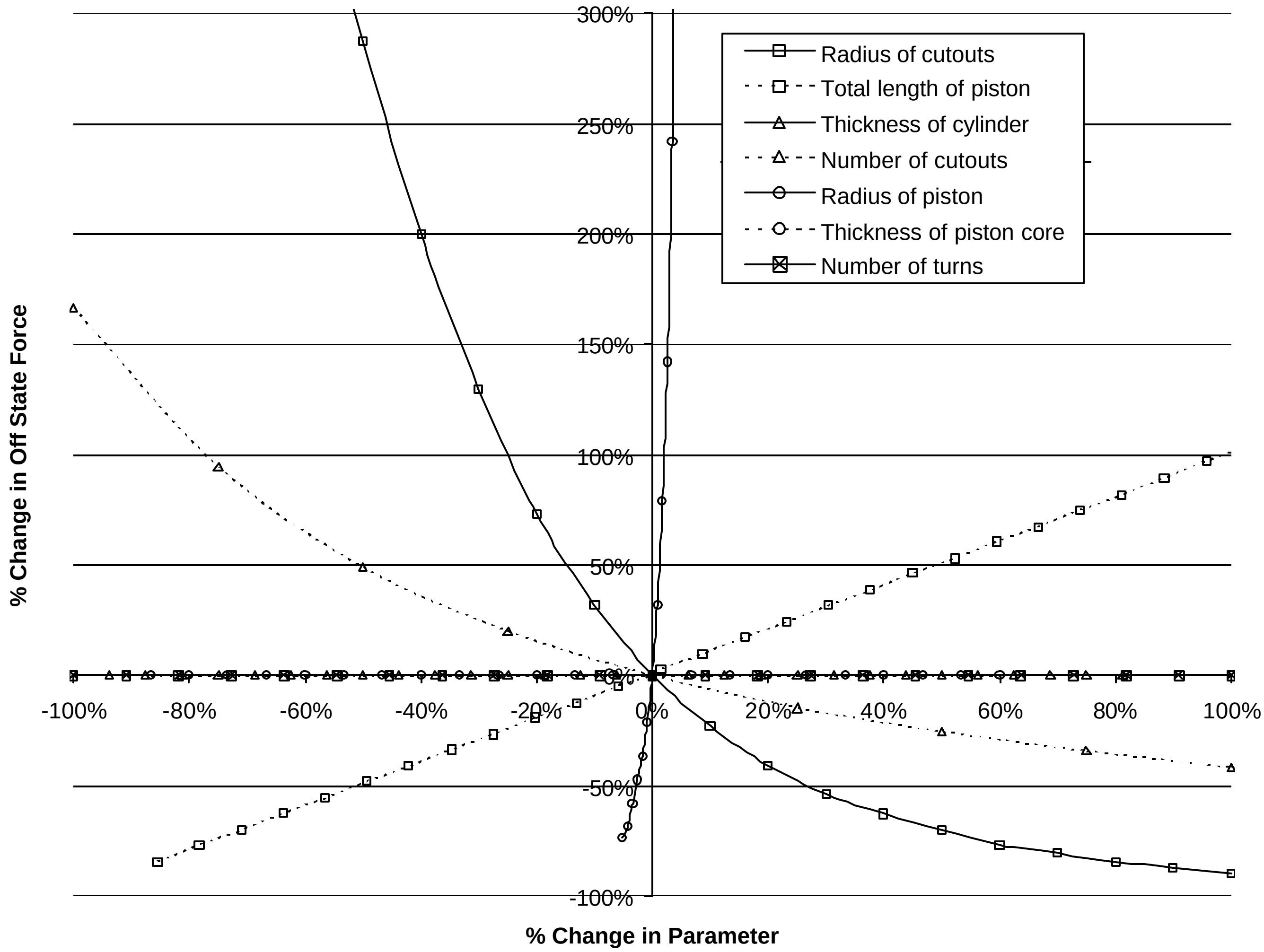

Figure 17. Sensitivity of off-state force to various parameters

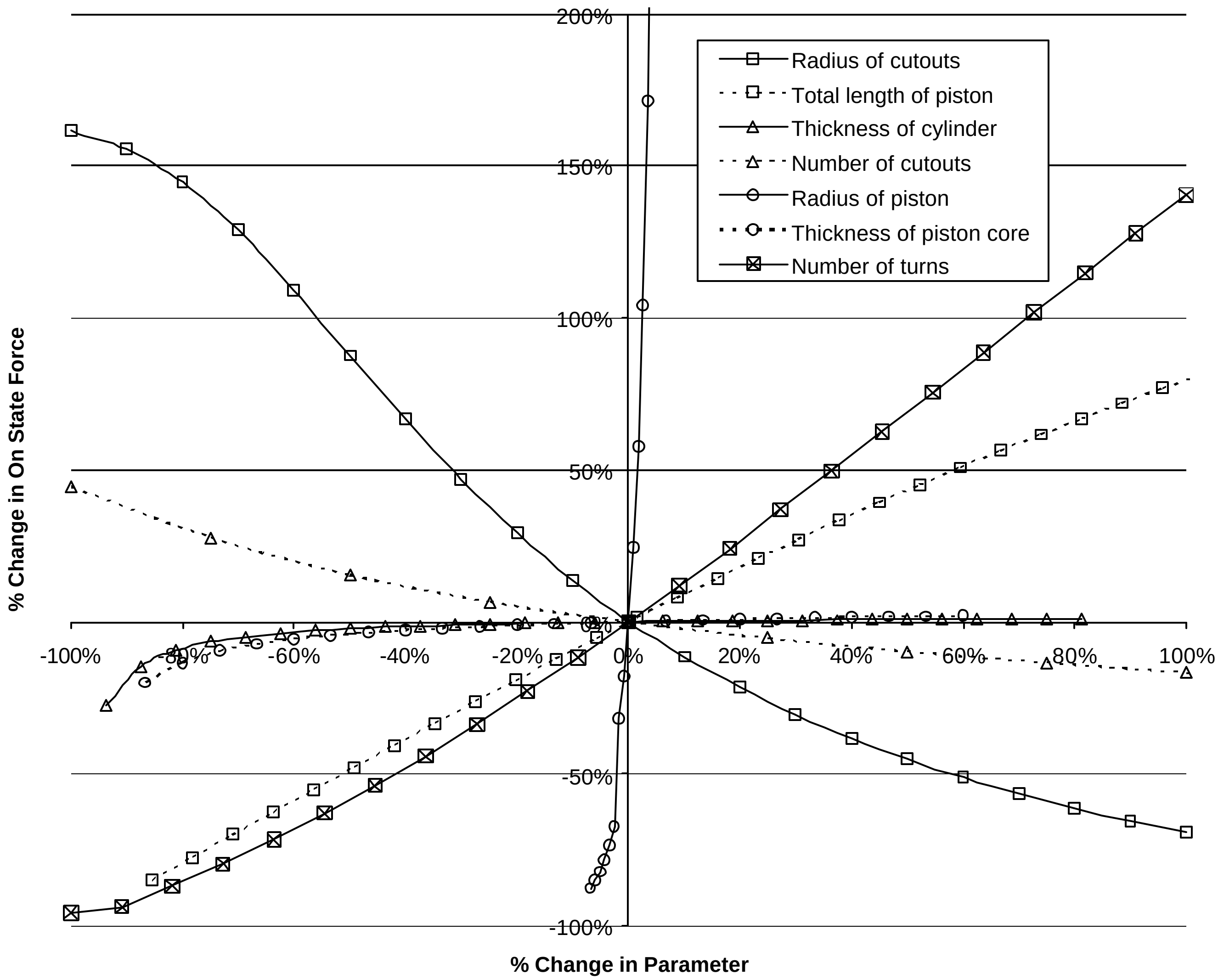

Figure 18. Sensitivity of on-state force to various parameters

Table 2. Operating points of sensitivity analysis

Parameter	**Operating Point**
Radius of cutout (20 cutouts)	2 mm
Length of piston	27.6 mm
Thickness of cylinder	4 mm
Number of cutouts (1 mm radius)	50
Radius of piston	30.6
Thickness of piston core	15 mm
Number of turns	275

From Figure 17 and Figure 18 it is clear that the forces are more sensitive to some parameters than others. The forces in both the on and off state vary linearly with the length of the piston. This can be expected because the flow rate is inversely proportional to the length of the piston, and the force is proportional to the flow rate (Chapter 2). As the length of the piston increases, both the on and off state forces increase. The radius of the piston is the most sensitive parameter, as varying it just slightly greatly affects the forces in both directions.

Looking at the off-state force as a function of the radius of the cutouts, it is clear that increasing the size of the cutouts greatly decreases the off-state force. From the on-state plot, it is seen that increasing the size of the cutouts reduces the on-state force as well, but not as drastically. As the radius of the cutouts gets large, the off-state force tends to level off and not decrease as much, while the on-state force continues to decrease almost as much as when the cutouts were small. In this case, varying the radius of 20 cutouts was investigated. Varying the radius of different numbers of cutouts yielded similar results.

As can be seen when comparing Figure 17 and Figure 18, the number of cutouts has more of an effect on the off-state force than the on-state force. In both states, the affect of the cutout is larger when the number of cutouts is small (in this case less than 20), and then somewhat levels out as the number of cutouts increases.

A few of the parameters explored only affect the on-state force because their role is solely in the magnetic circuit. These parameters do not affect the off state because there is no magnetic field in the off state. One such parameter is the thickness of the cylinder (assuming that the inner diameter remains constant, and the outer diameter is what changes). From Figure 18, it can be seen that the thickness of the cylinder does not have a large impact on the on-state force unless the cylinder is very thin, in this case less than 1 mm thick. When the cylinder is thicker than 1 mm, the on-state force is relatively constant. The force does increase slightly though, because increasing the cross sectional area of the cylinder decreases the reluctance of the cylinder in the magnetic circuit, and therefore increases the flux. Varying the radius of the piston core has the same effect as varying the cylinder thickness, as can be seen in both figures. The off-state force is constant, and the on-state force is relatively stable if the radius is larger than 8 mm. It was assumed that the number of coils was constant when varying the piston core, which is actually not very practical because there is less room for coils with a larger core. However, also seen from Figure 18, the effect of the number of coils is much more drastic than that of the piston core radius, so although increasing the radius of the piston core increases the on-state force, trading that space for coils is much more beneficial, provided that the core is above some minimum value. Increasing the thickness of the cylinder, the radius of the core, and the number of coils all increase the on-state force because they all increase the magnetic flux. However, increasing these past the point of magnetic saturation of the fluid will make no difference in the flux.

Based on the rough sensitivity analysis, it is apparent that some parameters are more sensitive than others. Also, parameters can be more sensitive around certain values or more sensitive in one state (on or off). Overall, the most sensitive parameter is the radius of the piston, and the least sensitive parameters are the thicknesses of the piston core and the cylinder. Since the goal of this project is to lower the off-state force, it is noted that the only parameters that affect the off-state force are the radius of the cutouts, the number of cutouts, the length of the piston, and the radius of the piston. Therefore, it is only necessary to vary these parameters to achieve the project objective. The length of the piston is not varied because it has the same effect in the off state as it does in the on state.

3.2 Design constraints

This analysis does not improve the entire design of a typical mixed mode damper; it just changes the space between the piston and cylinder by adding a different number and size of cutouts. If the entire design were to be improved, then the optimal thickness of the piston core, thickness of the cylinder, radius of the shaft, diameter of the wire, total length of the piston, length of the piston with coils, number of sections of coils, and the radius of the insulator would be determined as well. However, all of these parameters, except for the length of the piston, only affect the on state. They do not affect the off state, so they are not considered in this analysis. They are important parameters though, as they do affect the dynamic range. For instance, decreasing the thickness of the piston core allows more room for coils, which can increase the magnetic field, and therefore increase the on-state forces. However, the piston core needs to be thick enough for the magnetic flux path. Since the focus of this project is specifically to decrease the off-state force, this analysis just focuses on the radii of the piston and cutouts, and the number of cutouts.

3.3 Trends

The original intention of this project was to model the damper and then optimize the design using a formal optimization technique such as the optimization toolbox in MATLAB. However, since the model could not be completed analytically and had to be completed numerically, no formal optimization was conducted. Therefore, an evaluation of trends was done on the model.

Because the gap distance between the piston and the cylinder was determined to be the most critical parameter in the damper design, this evaluation concentrates on the radii of the piston and cylinder. The size and number of cutouts were also varied, as varying the size of cutouts essentially varies the piston radius. Everything else was held constant. In this analysis, -five different damper sizes were studied. The damper size is the cylinder radius, and for this analysis, the cylinder radii examined were 0.02, 0.04, 0.06, 0.08, and 0.10 meters. For each size, the radii of the piston and the semi-circular cutout as well as the number of cutouts were varied to see when there was benefit from the cutouts. Trends were then studied.

For a particular regular damper size, the piston radius was varied in 0.5 mm increments, and the on- and off-state forces were computed. For each regular damper explored, contoured-piston dampers were designed to match the on-state force, but to have lower off-state forces. In order to find a contoured-piston damper with a lower off-state force, the piston radius, cutout radius, and number of cutouts were varied. Every instance discovered in which a contoured-piston damper had a lower off-state force than a regular-piston damper, but with an equivalent on-state force, was declared an instance of benefit. The amount of benefit was quantified by a factor referred to as gain, which is the improvement in the off-state force divided by the original off-state force. The improvement in the off-state force is the regular piston off-state force minus the contoured piston off-state force. The instance was only considered to be one of benefit if the gain was larger than 5%.

For each damper size, different piston sizes were studied. Piston radii examined were taken in 0.5 millimeter increments, the largest being 0.5 millimeters smaller than the cylinder radius and the smallest being the point at which no benefit was found, which was different for each damper. For each piston size, the number of semi-circular cutouts was varied as well as the radius of the cutouts. For each sized cutout, the minimum number of cutouts examined was one, and the maximum number of cutouts was the number of cutouts it took to fill the entire piston circumference with cutouts.

The goal of studying all the instances of gain was to identify trends in the size and number of cutouts and the size of the piston that produced gain. Analyzing all the instances of gain, relationships were identified between the radius of the piston, radius of the cutout, and radius of the cylinder of the piston in order to note trends. Many strong correlations were noted. For each instance of benefit, the magnitude of the gain was quantified. Also, for each instance of gain other quantities were noted such as the fraction of the contoured piston that was covered with cutouts and the ratios between the radii of the cutout, piston, and cylinder (Figure 19).

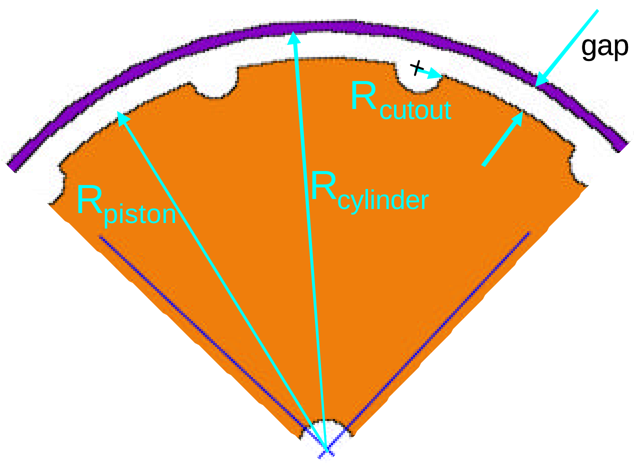

Figure 19. Labeling of parameters

The first strong correlation found involved the regular-piston damper that could be improved with cutouts. $R_{regular}$ is the radius of the regular piston that has the desired on-state force, but the off-state force is too high. Figure 20 is a plot of the ratio between $R_{regular}$ and $R_{cylinder}$ versus gain. Each marker on the plot represents an instance of benefit, where a contoured piston damper had a lower off-state force but equivalent on-state force as a regular-piston damper. Designs that did not give benefit are not represented on this plot. Both the number and size of the cutouts are varied in this plot. It can be seen that there is a lot of potential gain. The optimal range for dampers whose off-state force can be lowered with cutouts have a piston radius that is 94-97% of the cylinder radius (Figure 20). A piston much larger than 97%

of the cylinder does not provide many opportunities for benefit because both the on- and off-state forces are so large that cutouts would not significantly decrease the off-state force without significantly decreasing the on-state force as well. A piston much smaller than 94% of the cylinder does not provide many opportunities for benefit because the off-state force is already so low that cutout would have more of an effect on the on state than on the off state.

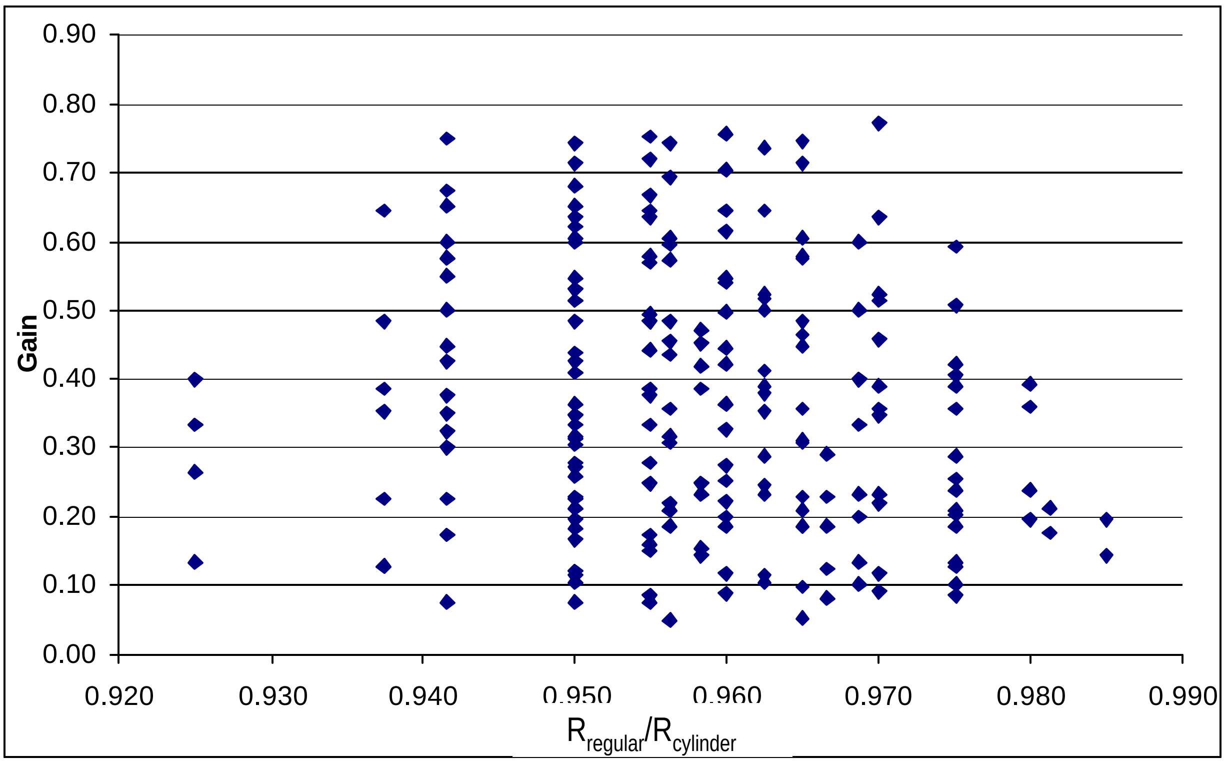

Figure 20. Correlation between $R_{regular}/R_{cylinder}$ and gain

Another strong correlation has to do with the ratio of piston radius and cutout radius with the percent gain (Figure 21). The percent gain is highest when the ratio between the cutout radius and the piston radius is between 5 and 10%, but there are many more instances of gain when the ratio is between 1 and 5%. This implies that numerous dampers can benefit from small cutouts, and fewer dampers can benefit from larger cutouts, though the gain could be higher. In the 5 to 10% region, most of the dampers are larger, while the 1 to 5% range contains dampers of all sizes. There is almost no gain when the radius of the cutout is less than 1% or greater than 15% of the piston radius.

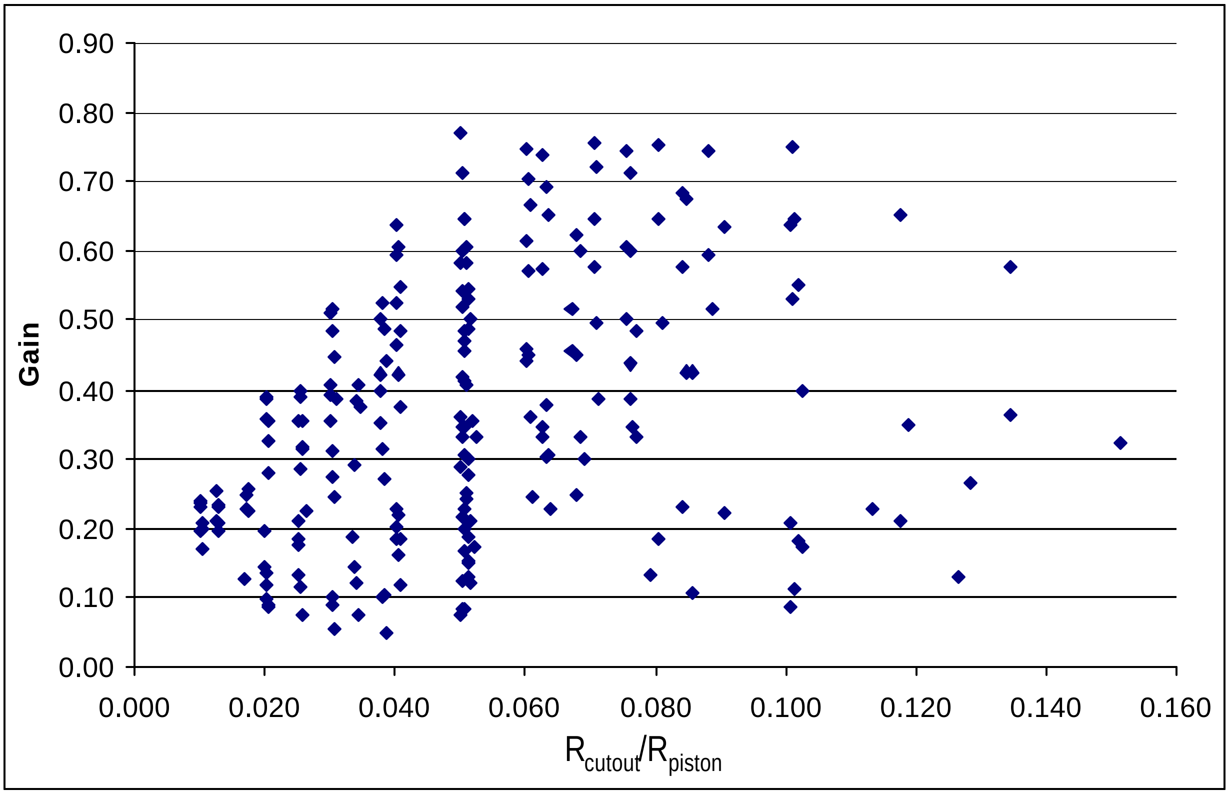

Figure 21. Correlation between R_{cutout}/R_{piston} and gain

A third correlation has to do with the amount of the piston that was covered with cutouts. It can be seen that there is a lot of potential gain when around 20-40% of the piston is covered with cutouts. There is an even higher potential for gain when the piston is 90-99% covered with cutouts. Dampers that benefited from the 20-40% coverage region tended to be only larger dampers, and have a small number of fairly large cutouts. On the other hand, dampers in the 90-99% coverage region were of all sizes, and had small cutouts that covered nearly the entire piston circumference. The minimum percent coverage that yielded gain was 13%, and the maximum was 99%.

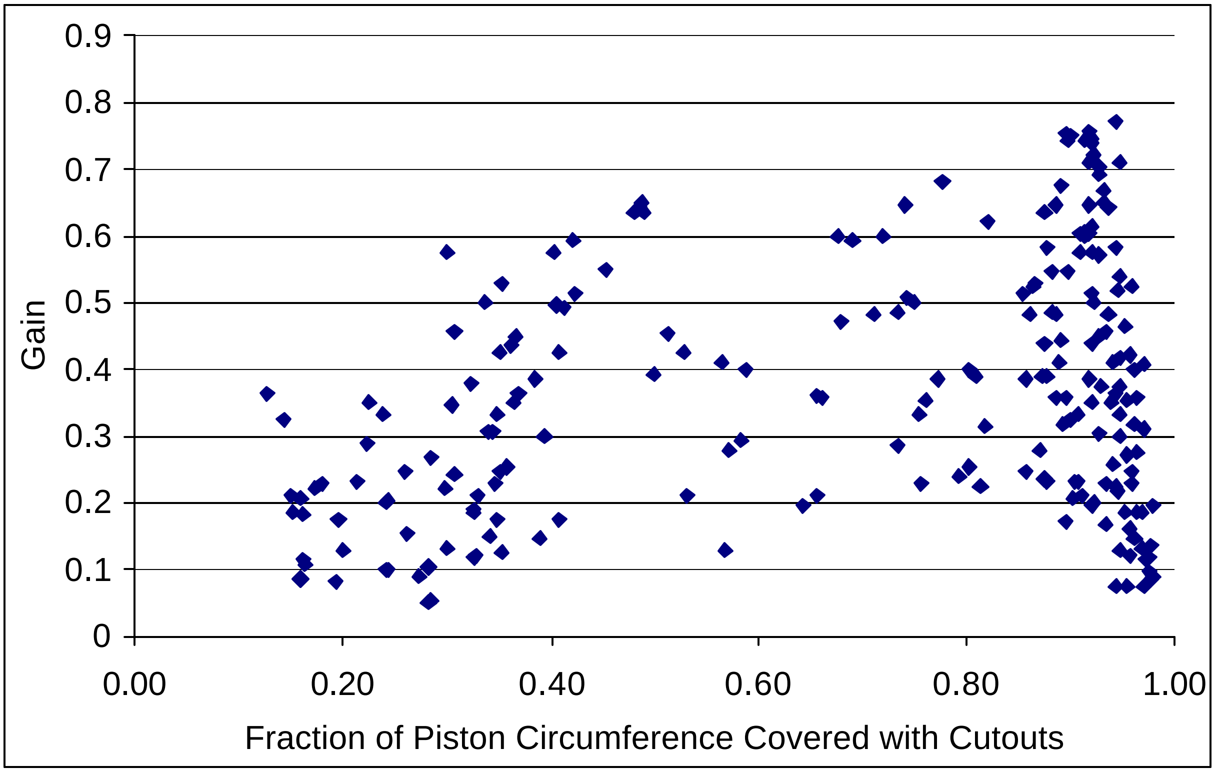

Figure 22. Correlation between fraction of piston covered with cutouts and gain

4. Materials and Apparatus

Once the design was analyzed, a damper design was selected to be built. The design chosen produces a lower off-state force while maintaining the same on-state force as a regular damper of the same size. The comparison damper was not built, but may be in the future. The two dampers, the contoured piston damper and the regular damper, are identical except for their pistons.

In designing the prototype damper, the first thing that was determined was the desired force range. The requirements were an on-state force of at least 5 kiloNewtons at a current of 1.2 amps and a piston velocity of 0.1 meters/second, and an off-state force as small as possible. It was also required that the radius of the cylinder be 31.75 millimeters (1.25 inches), as that was a standard available tube size. The radius of the shaft was set to be 12 millimeters because it was a standard available size. The length of the piston was set to 27.6 millimeters. The regular-piston damper that meets these specifications has a radius of 30.24 millimeters, and on-state force of 5.4 kiloNewtons, and an off-state force of 37 Newtons. This would make the ratio between the radius of the regular piston and the cylinder 95.4, which fits in the optimal range for dampers whose off-state forces can be lowered with cutouts (Figure 20).

Next the size of the cutouts needed to be determined. Since the damper being designed is relatively small compared to the dampers examined in the study, the most benefit would be when the radius of the cutout was around 3 to 4% of the radius of the piston (Figure 21). It was decided that the radius of the cutout would be 1 millimeter, which is 3.3% of the piston radius. Then, based on Figure 22, it was decided that the entire piston circumference would be covered with cutouts since that provides the most benefit with smaller dampers. Based on these conditions, the outer radius of the piston was varied until the on-state force requirement of 5.4 kiloNewtons was reached. The piston radius was calculated to be 30.6 millimeters. Therefore, the gap between the piston and the cylinder on the contoured piston damper varies from 1.15 to 2.15 millimeters; the gap in the regular damper is a constant 1.26 millimeters. With a piston radius of 30.6 millimeters, 96 cutouts fit around the circumference of the piston (Figure 23).

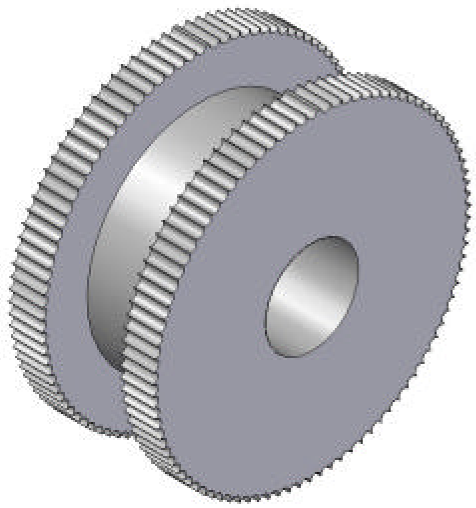

Figure 23. Contoured piston

The contoured-piston damper has a calculated off-state force of 16 Newtons, which is 57% reduction in off-state force. The dimensions for the contoured piston damper are given in Table 3. Table 4 contains all of the specifications of the parts of the prototype damper.

Table 3. Dimensions of damper built

Parameter	Variable Name	Value
Radius of piston (mm)	$R_{regular}$ / R_{piston}	30.6
Radius of cylinder (mm)	$R_{cylinder}$	31.75
Shape of cutouts		Semi-circle
Number of cutouts	n	96
Radius of cutouts (mm)	R_{cutout}	1
Radius of shaft (mm)	R_{shaft}	12
Diameter of wire (mm)	d_{wire}	0.375
Total length of piston (mm)	L	27.6
Length of piston without coils (mm)	$L_{noncoils}$	12.6
Length of piston with coils (mm)	L_{coils}	15
Radius of piston core (mm)	R_{pcore}	15
Number of coils	N	500
Thickness of cylinder (mm)	t	4
Number of sections of coils	N_S	1
Radius of insulator (mm)	$R_{insulator}$	9

Table 4. Component specifications

	Material	Manufacturer
Piston	1018 steel	
Insulator	Aluminum	
Shaft	Stainless steel	
Cylinder	1018 steel	Tubular Techniques
Plug	Aluminum	
End Plate	Aluminum	
Winding	Copper	
Seals	Polymer	Chesterton
Fluid	MRF-132AD	Lord Corporation

The piston and cylinder in the damper must be made of material that will conduct the magnetic field. Low-carbon steels work well for the design and in terms of availability, so 1018 steel was used. A hole was placed in one piston end disc as close to the outer edge as possible. The hole allowed a spot for the wire to exit the piston before it is brought back up the shaft.

The insulator had to be made of a material that would not conduct the magnetic field in order to concentrate the field on the fluid. Therefore aluminum was used for this part. The insulator was press fitted into the piston, and the shaft was press fitted into the insulator. The top of the insulator was also pinned to the shaft. Since the insulator sticks out of the piston on the top but not on the bottom, during testing the piston will only be run in one direction.

The shaft, as was the case with the insulator, had to be made of a material that would not conduct the magnetic field, so it was stainless steel. A hole in the axial direction extended from the top of the shaft and went down through about midway through the shaft. The hole allowed a way for the wires to get back

The purpose of the plug (Figure 24) is to confine the fluid inside the cylinder. A ¾”-thick plug fits inside the cylinder at each end so that it is flush with the cylinder. Leakage of the fluid between the plug and the cylinder tends to be a problem in mixed mode dampers, so a strong bond is required between the plug and cylinder. Therefore, the plug will be semi-permanently sealed inside the cylinder using Loctite. The shaft goes through the center hole with a running fit. A seal fits around the shaft and sits in the slot. The three holes around the seal are threaded holes for securing the end plate on to the plug. The hole at the bottom of the plug is used to fill the damper with fluid. Only one of the plugs has this hole. Once the damper was filled, the hole is covered with an NPT plug.

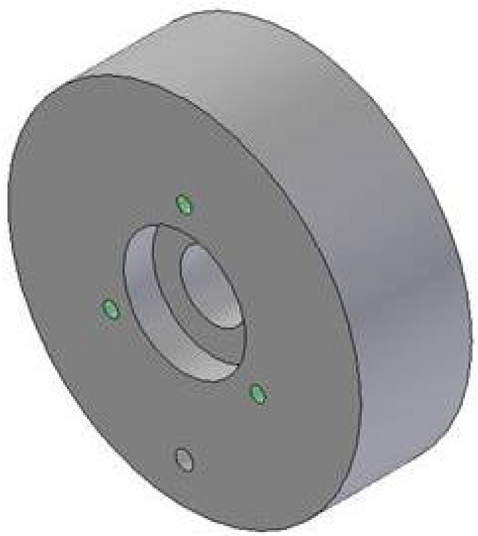

Figure 24. Solid Edge model of the plug

The purpose of the end plate (Figure 25) is to secure the seal in place. The center hole is large enough to comfortably slide the shaft through, but small enough to press down on the seal and hold it in place. The three through holes along the edge of the end plate are to screw the end plate to the plug. There are two identical end plates, one for each end of the damper.

Figure 25. Solid Edge model of the end plate

Wire is wrapped in layers around the piston core to create the magnetic field. The wire is 26 gage copper wire. As mentioned previously, there is one seal at each end of the damper. The seal (Figure 26) used is a Chesterton 5K Super Wiper. Though the seal has a tight fit on the shaft, the seal allows the shaft to glide through the plug by keeping it centered. The tight fit helps prevent fluid leakage.

Figure 26. Seal from Chesterton

The fluid used for this project is MRF-132AD, which is manufactured and sold by the Lord Corporation. The fluid is a hydrocarbon-based fluid with a viscosity of around 0.09 Pa-s. For this project, the viscosity is a very important property of the fluid because the off-state force is largely a function of the viscosity. Regardless of the fluid chosen, the model predicts that the contoured piston will have a 56% lower off-state damping force than its comparable regular damper. Although the percent is the same, the magnitude of the benefit varies greatly with the viscosity. If the fluid used had been MRF-240BS, a water-based fluid with a viscosity of 5.0 Pa-s at high shear rates, the predicted magnitude off-state force of the contoured piston damper would be over 1 kiloNewton lower than that of the regular damper. The yield stress of the fluid as a function of the magnetic field strength can be seen in Figure 27. MRF-132AD can achieve a yield stress of around 40 kPa at a magnetic field strength of 250 kAmp/m before it becomes magnetically saturated. Out of the 500 milliliters of fluid in the damper, only about 4 milliliters are active at a time when the magnetic field is on.

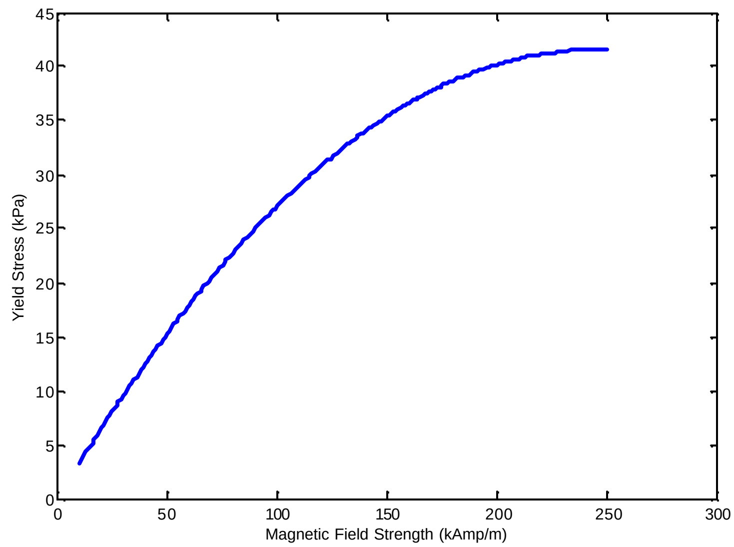

Figure 27. Yield stress property of fluid MRF-132AD

Based on these results, it is apparent that this damper would be particularly useful for high-load applications. The on-state force is designed to reach 5.4 kiloNewtons at a current of 1.2 amps, and the off-state force is designed to be around 16 Newtons. This suggests that a good application of this damper would be in applications requiring large damping forces, such as in civil structures.

5. Conclusions and Future Work

5.1 Conclusions

In conclusion, MR dampers can be useful in many applications, ranging from bridges to heavy vehicle seating. Mixed mode is a common MR damper design mode with a simple design, that is capable of producing very high on-state forces, but is somewhat limiting with its high off-state forces. The research done for this thesis found that the off-state force in mixed mode MR dampers can be reduced by putting cutouts around the piston circumference. The number and size of the cutouts depends upon the size of the damper, the force desired, and other constrained parameters (such as the length of the piston).

Trends of when dampers could benefit from cutouts in the piston were examined. These trends suggested that in many dampers could have lower off-state forces with equal on-state forces by putting cutouts in the piston. Larger dampers can benefit from a small number of fairly large cutouts or from a large number of very small cutouts. Small dampers tend to benefit more from many small cutouts.

5.2 Future Work

In this design analysis, many simplifying assumptions were made. One significant assumption was that the magnetic flux lines traversed the gap radially. However, magnetic flux tends to concentrate around sharp edges and corners. Since the damper has semi-circular cutouts around its circumference, it is likely that this concentration would significantly change the yield stress. This should be taken into account, since an increase in yield stress around the cutouts could greatly benefit the design.

It is also recommended that the design analysis be taken a step further such that the design for any regular mixed mode MR damper can be computed. The parameters, including size constraints and damping forces desired, would be in the inputs. The outputs would be a piston design with a specified piston radius, cutout radius, and number of cutouts.

A final recommendation for future work would be to optimize the entire design of a mixed mode damper to maximize the dynamic force range. The analysis completed in this work focused on lowering the off-state forces, and did not conduct any sort of optimization. However, work should be done on simultaneously increasing on-state forces and decreasing off-state forces. Also, other cutout geometries should be explored.

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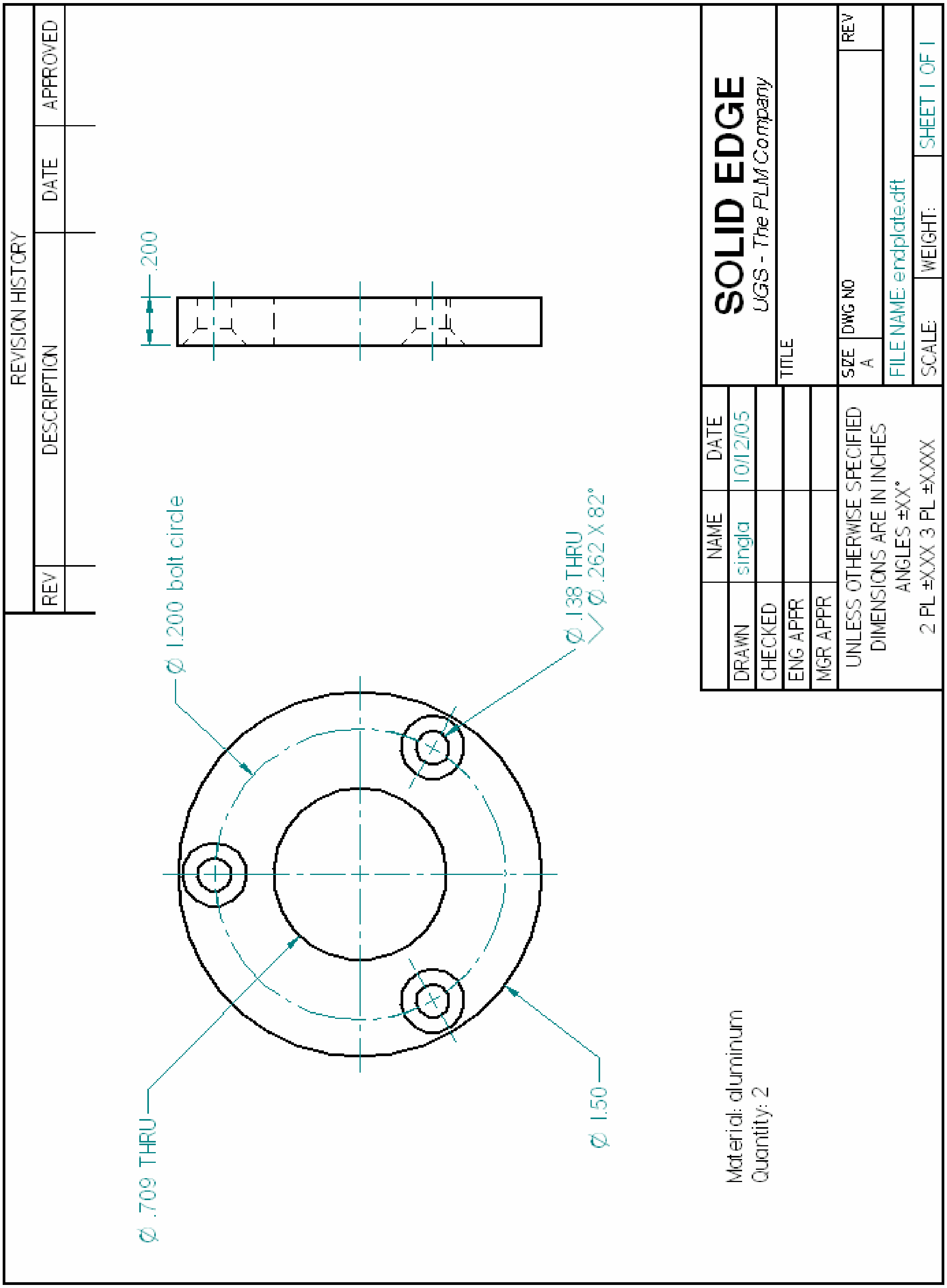
Appendix

Solid Edge drawings of prototype damper parts


REVISION HISTORY
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DATE
APPROVED
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.200
Ø .138 THRU
⌵ Ø .262 X 82°
Ø .709 THRU
Ø 1.50
Material: aluminum
Quantity: 2
NAME
DATE
DRAWN
singla
10/12/05
CHECKED
ENG APPR
MGR APPR
UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
ANGLES ±XX°
2 PL ±XXX 3 PL ±XXXX
SOLID EDGE
UGS - The PLM Company
TITLE
SIZE
A
DWG NO
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SCALE:
WEIGHT:
SHEET 1 OF 1

REVISION HISTORY			
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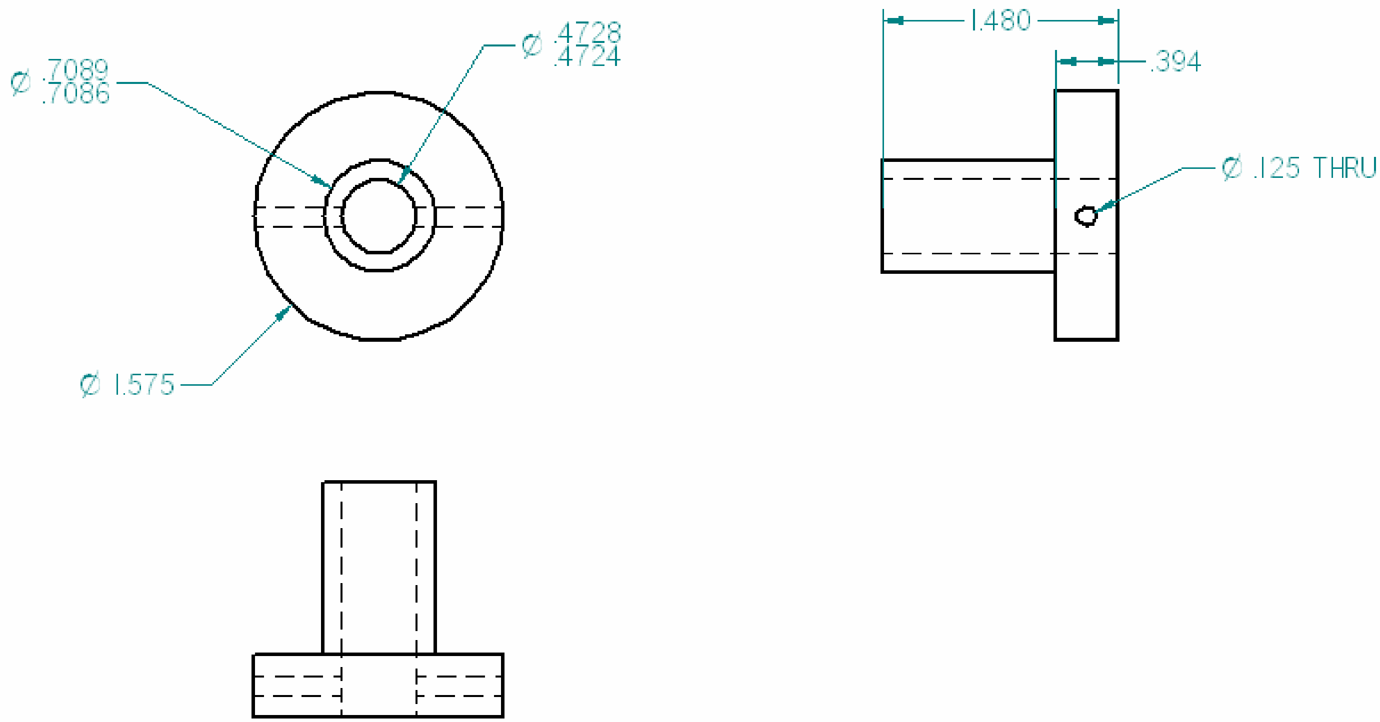

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CHECKED		
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MGR APPR		

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
ANGLES ±XX°
2 PL ±X.XX 3 PL ±X.XXX

SOLID EDGE
UGS - The PLM Company

TITLE

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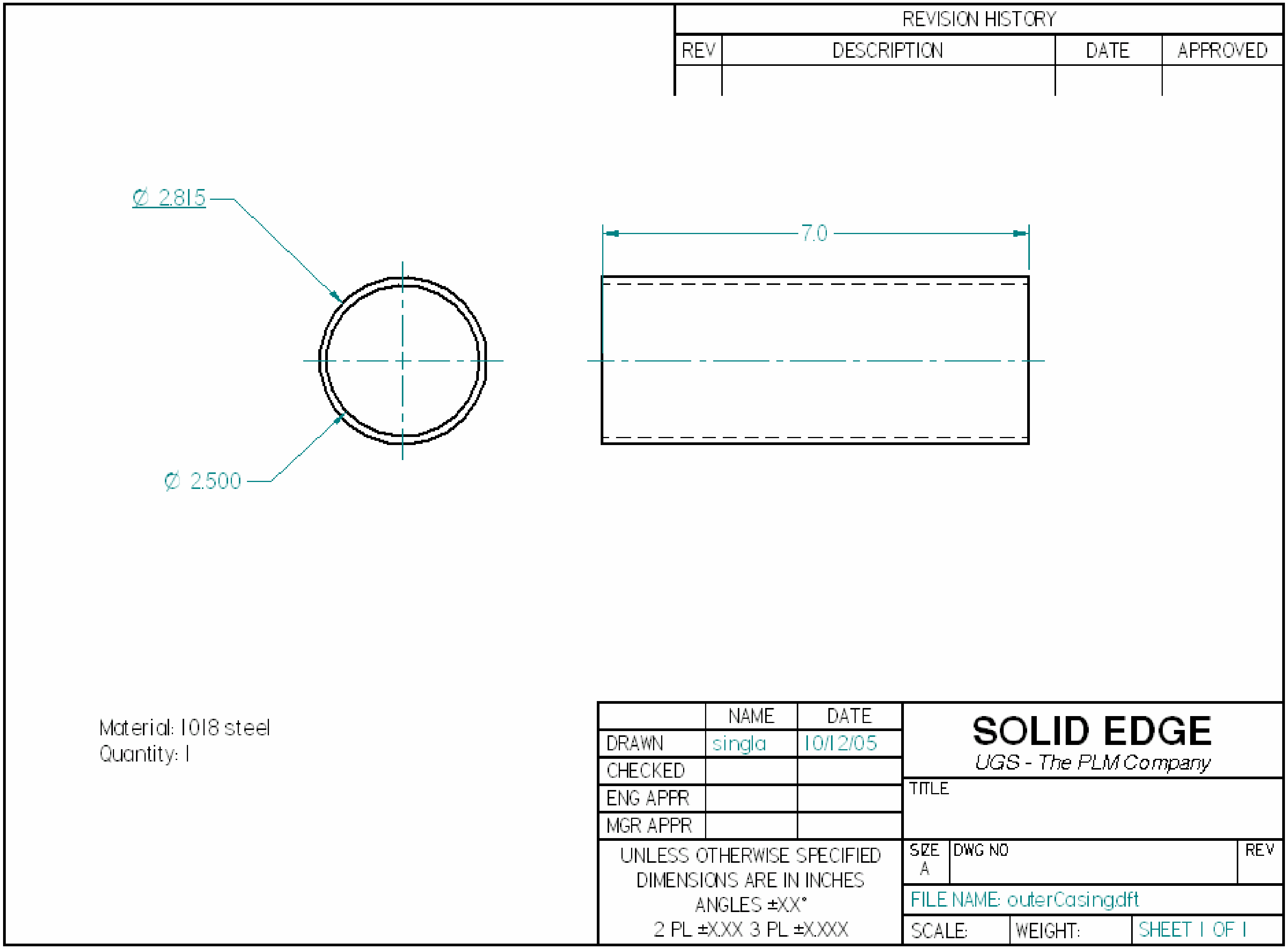
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DATE
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Ø 2.500
7.0
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DATE
DRAWN
singla
10/12/05
CHECKED
ENG APPR
MGR APPR
UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
ANGLES ±XX°
2 PL ±X.XX 3 PL ±X.XXX
SOLID EDGE
UGS - The PLM Company
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SIZE
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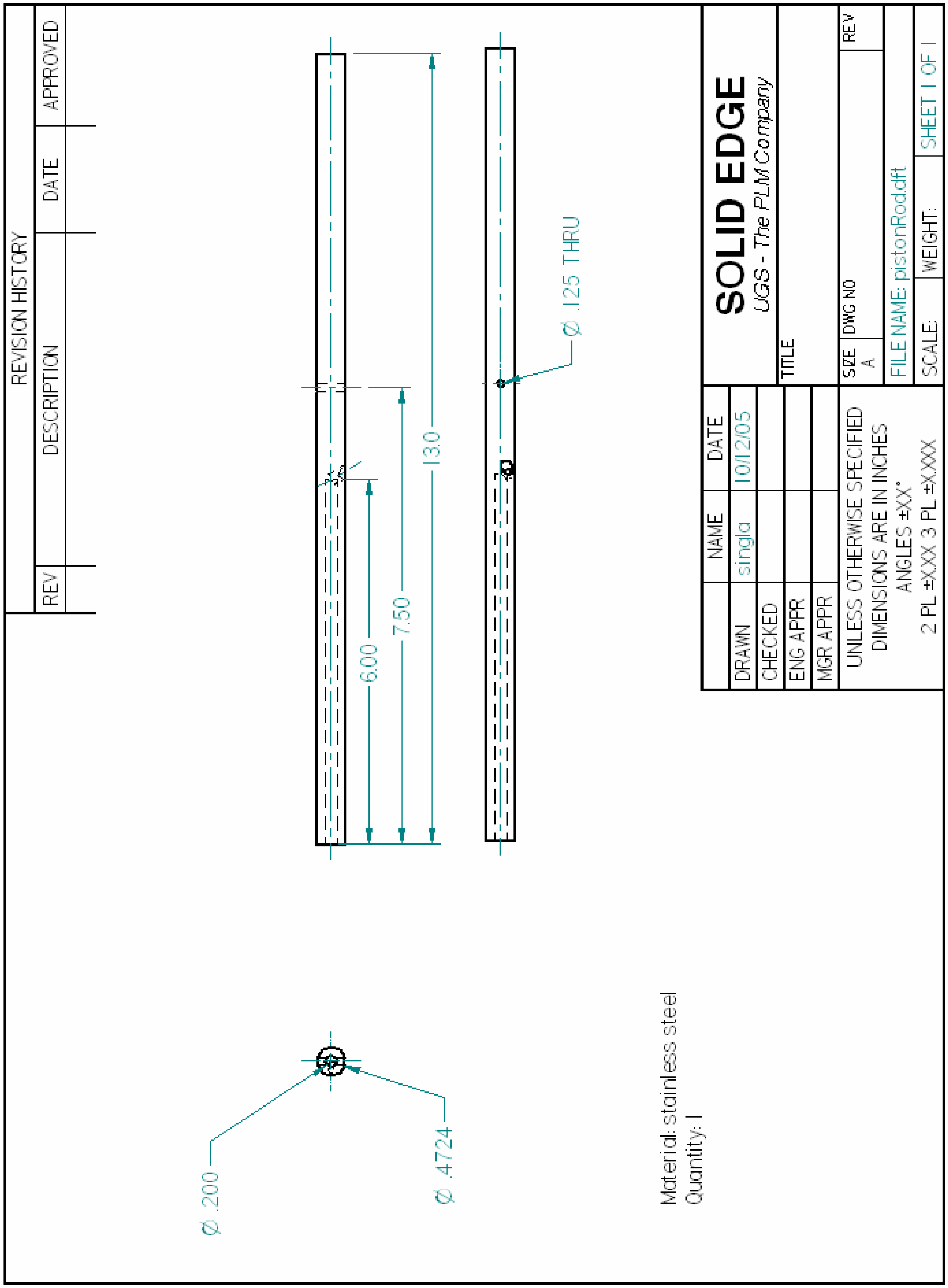
WEIGHT:
SHEET 1 OF 1


REVISION HISTORY
REV
DESCRIPTION
DATE
APPROVED
13.0
7.50
6.00
Ø .125 THRU
Ø .200
Ø .4724
Material: stainless steel
Quantity: 1
SOLID EDGE
UGS - The PLM Company
NAME
DATE
DRAWN
singla
10/12/05
CHECKED
ENG APPR
MGR APPR
UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
ANGLES ±X.X°
2 PL ±X.XX 3 PL ±X.XXX
TITLE
SIZE
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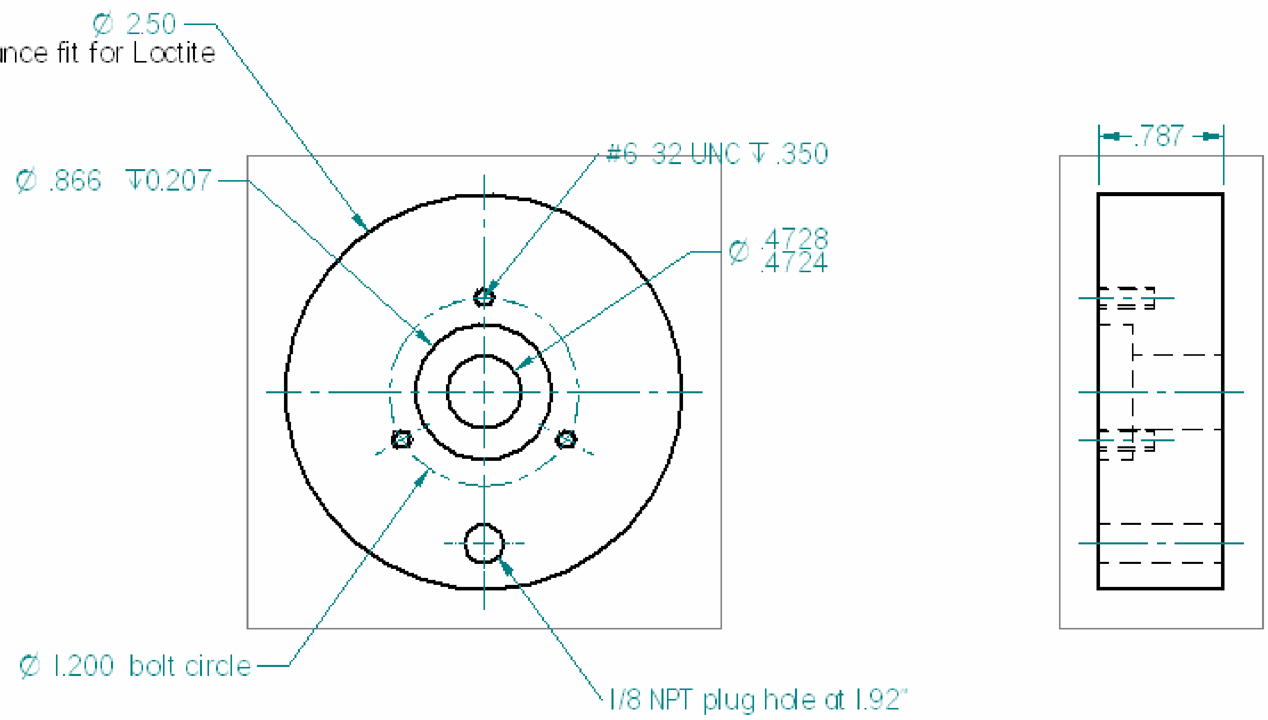
SCALE:
WEIGHT:
SHEET 1 OF 1

REVISION HISTORY			
REV	DESCRIPTION	DATE	APPROVED



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CHECKED		
ENG APPR		
MGR APPR		

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
ANGLES ±XX°
2 PL ±X.XX 3 PL ±X.XXX

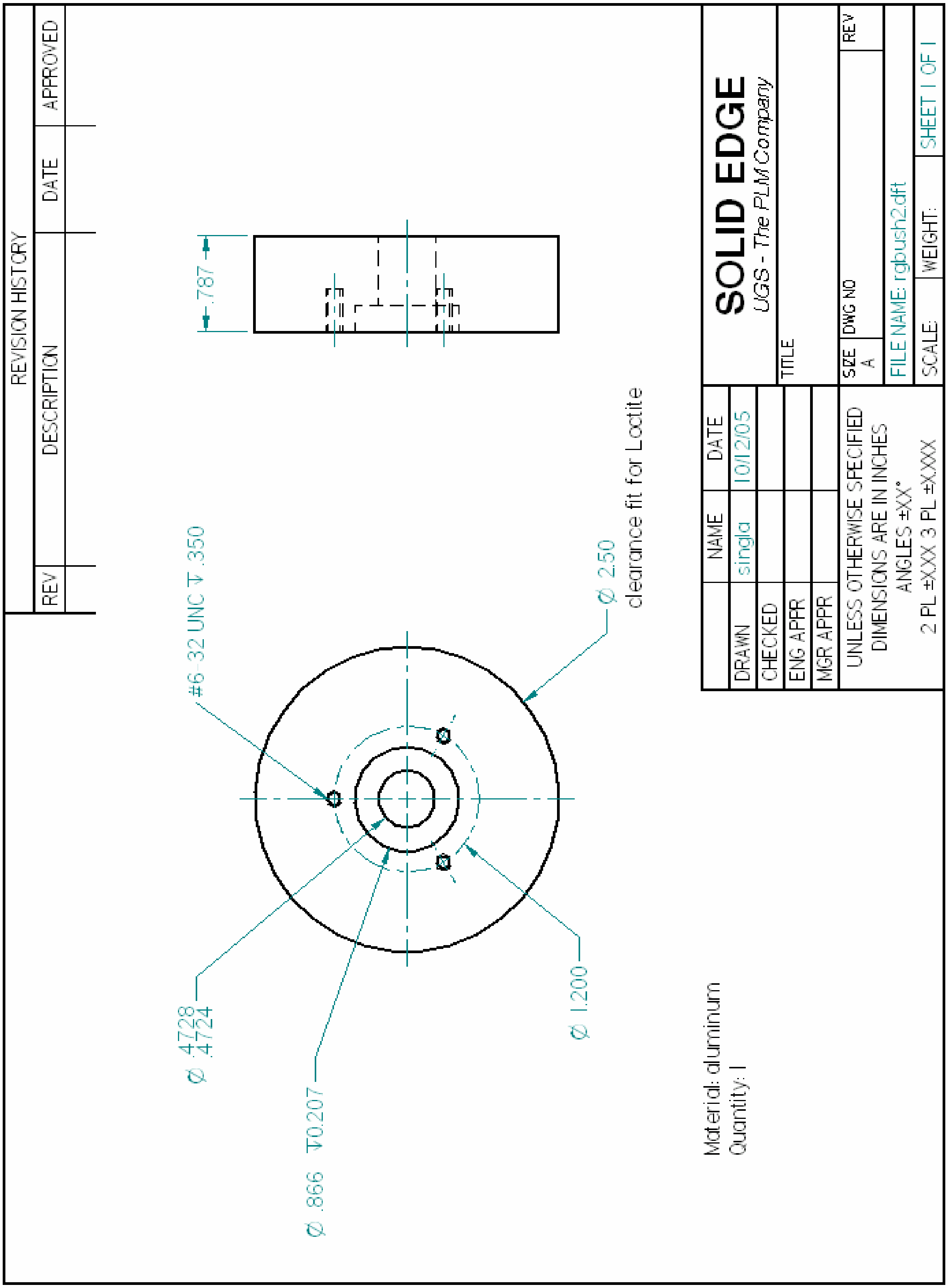
SOLID EDGE
UGS - The PLM Company

TITLE

SIZE	DWG NO	REV
A		

FILE NAME: rgbushdft

SCALE:	WEIGHT:	SHEET 1 OF 1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REV
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DATE
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clearance fit for Loctite
Ø .4728
.4724
Ø .866 ⊽0.207
Ø 1.200
Material: aluminum
Quantity: 1
NAME
DATE
DRAWN
singla
10/12/05
CHECKED
ENG APPR
MGR APPR
UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
ANGLES ±XX°
2 PL ±XXX 3 PL ±XXXX
SOLID EDGE
UGS - The PLM Company
TITLE
SIZE
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DWG NO
REV
FILE NAME: rgbush2.dft
SCALE:
WEIGHT:
SHEET 1 OF 1